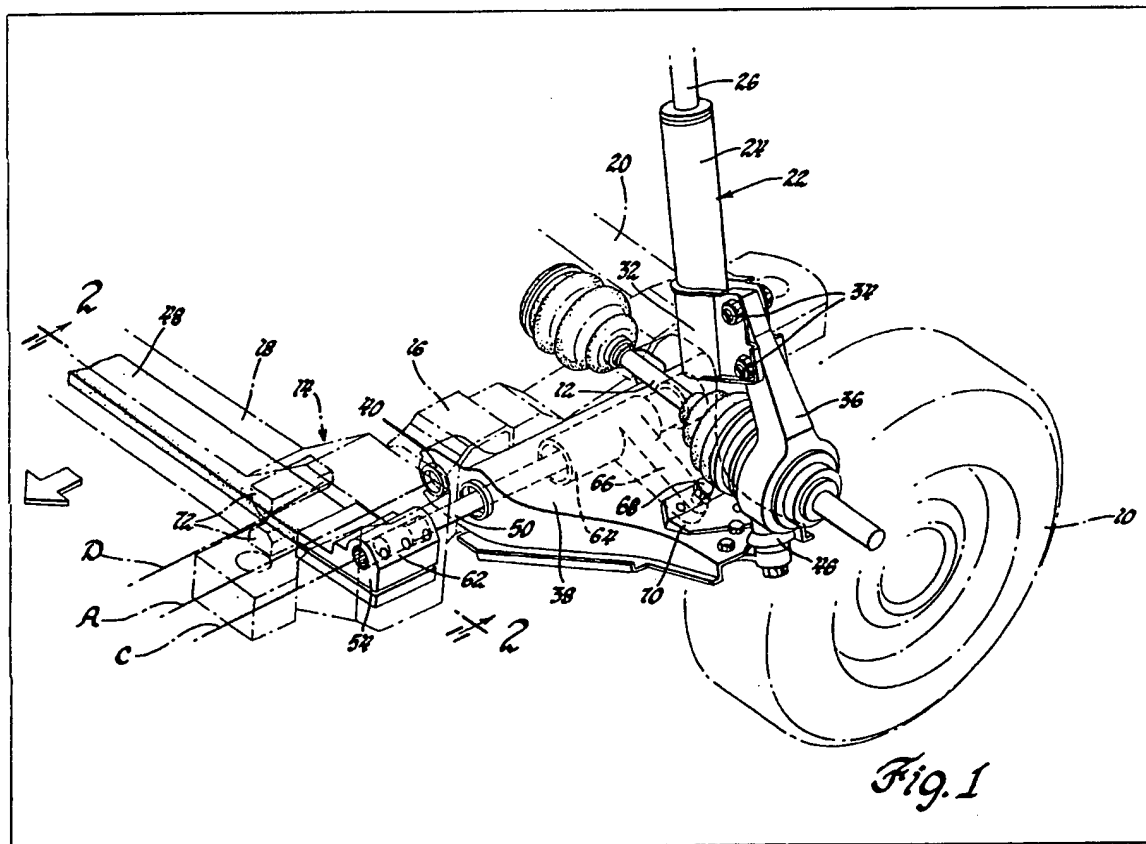


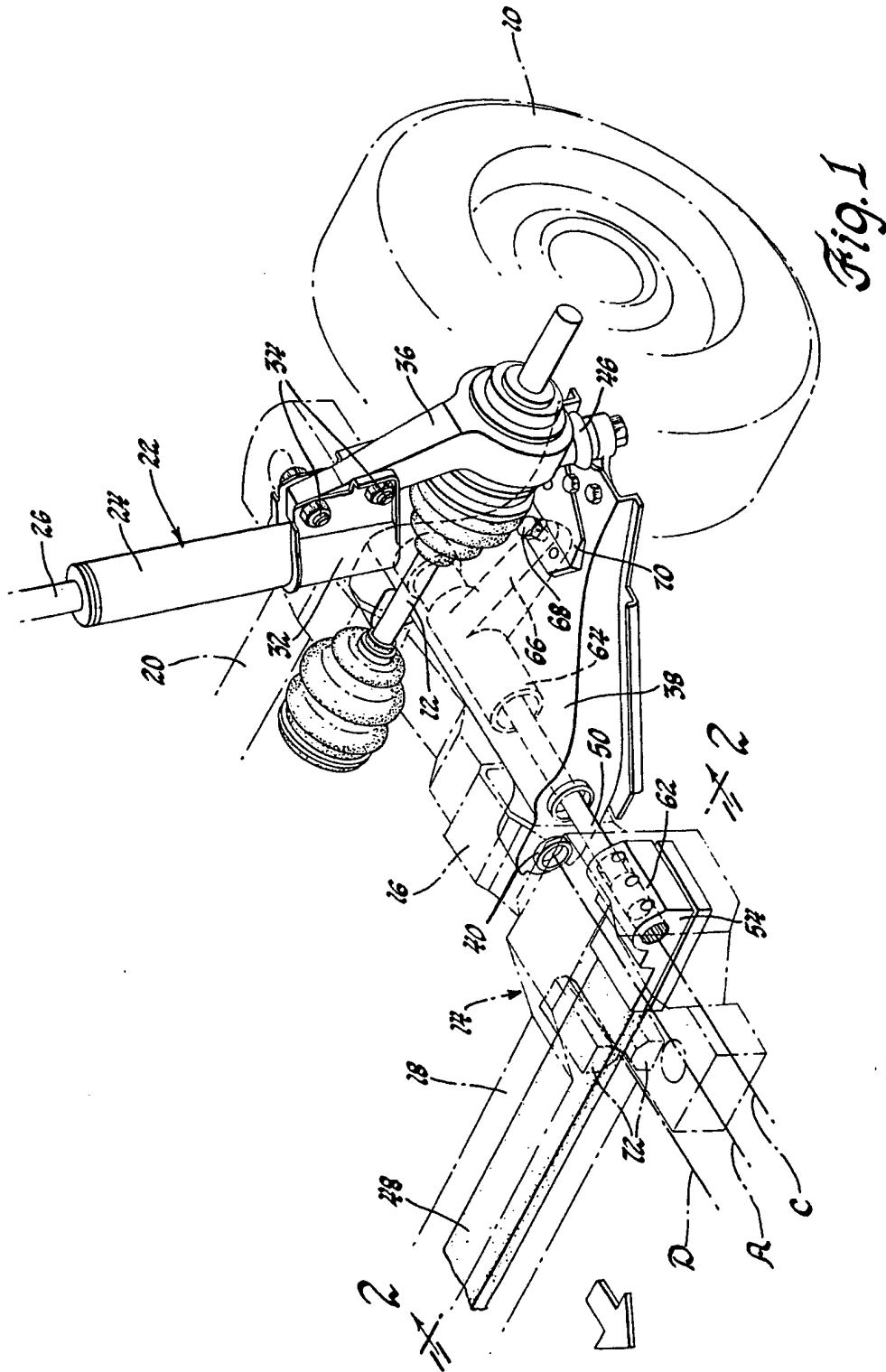
- (57) A vehicle suspension system includes transverse control arms 38 for an opposed pair of road wheels 10 and a primary suspension spring beam 48 spaced in a direction longitudinally of the vehicle from the control arms and interconnected therewith by torsion rods 50 in-

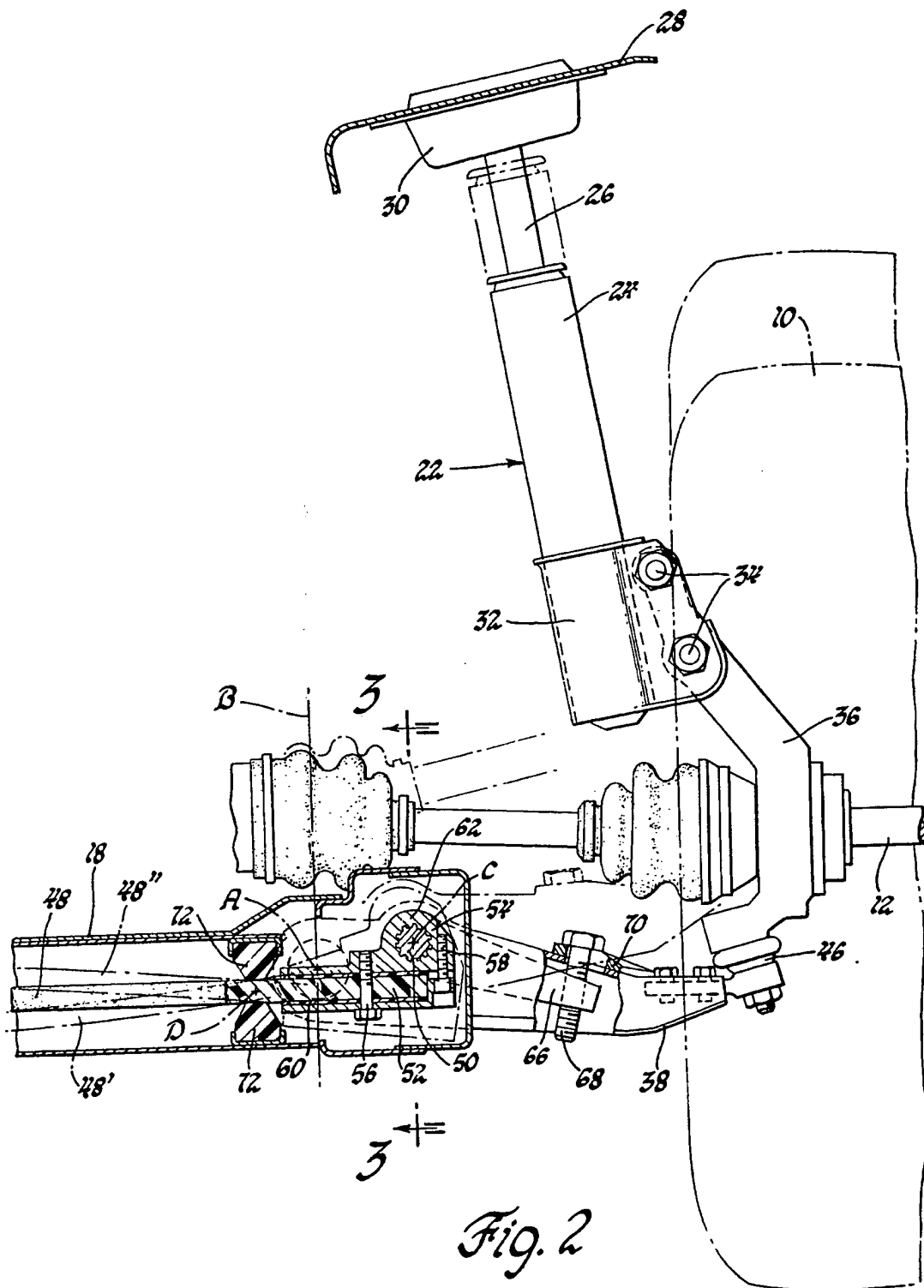
stalled on axes C located outboard of the swing axes A of the transverse control arms. The arrangement affords greater latitude to the vehicle designer in both space utilization and body styling.



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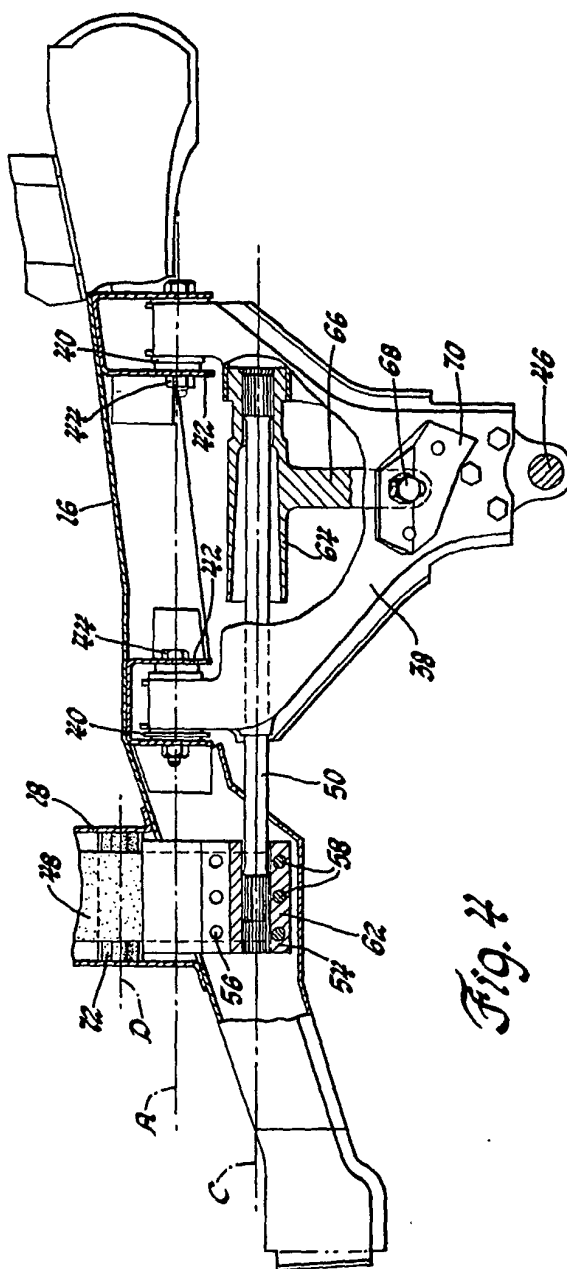
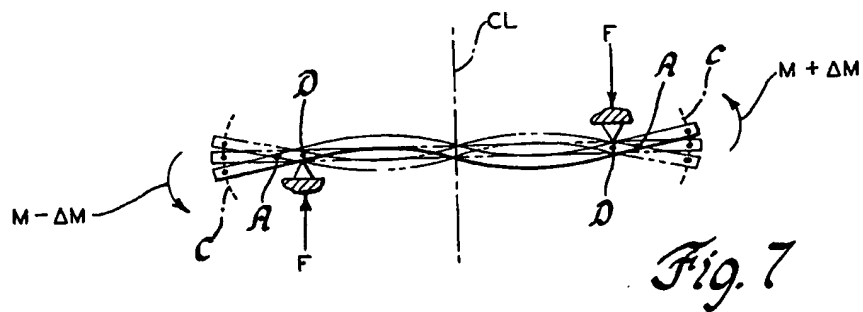
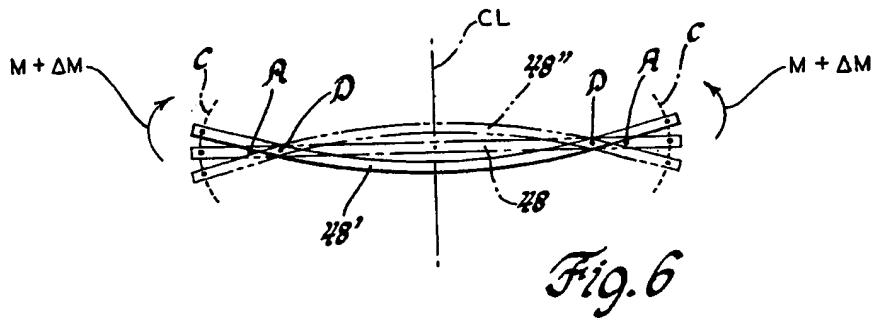
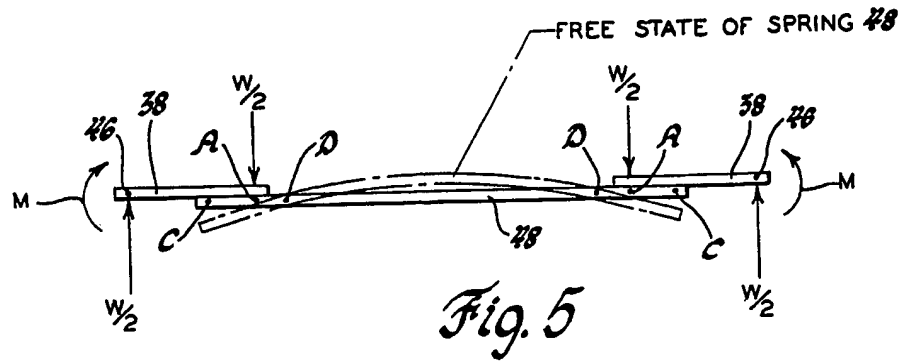


Fig. 2.



SPECIFICATION

Vehicle suspension system

- 5 This invention relates to vehicle suspension systems.

The installation of a total drive aggregate in either a front or rear compartment of the modern vehicle, combined with the general trend towards reduction of volume and mass in those areas of the vehicle body, imposes substantial design limitations in the use of conventional chassis systems, particularly vehicle suspension elements. Space and structure formerly available for locating such items as suspension control arms, vertical shock struts, coil springs and the like is now needed for drive train components. Moreover, modern combined spring and shock absorber strut suspensions of the so-called MacPherson type provide no completely satisfactory answer since they dictate high hood and quarter profiles resulting from the vertical stacking of the suspension elements typical of such arrangements.

The present invention is concerned with a vehicle suspension system affording greater latitude to the vehicle designer in both space utilization and body styling.

By the present invention there is provided a suspension system for the sprung mass of a vehicle relative to transversely opposite pair of road wheels, comprising a pair of transverse control arms each adapted to carry a respective one of the wheels and swingably mounted at a respective side of the sprung mass on a generally longitudinally extending axis thereof, a primary suspension spring comprising an elastic beam extending transversely of the sprung mass proximate the control arms but spaced from the control arms in a direction longitudinally of the sprung mass and having end portions projecting outboard of generally vertical planes containing the swing axes of the control arms, means connecting each control arm to a respective end portion of the elastic beam at a location on the latter outboard the respective said generally vertical plane and operative to carry to the beam forces effective to deflect the control arms relative to the sprung mass, whereby under the action of such forces the beam is adapted to bend about a spaced pair of nodal points located substantially inboard of the connection locations, and reaction mount means on the sprung mass engaging the elastic beam at the nodal points thereof.

Thus this suspension system includes a pair of transversely oriented control arms for the opposed road wheels, adapted to swing about longitudinal axes of the sprung mass in conventional manner. However, rather than connecting with such control arms a primary suspension coil or like spring at each side in the vicinity of each road wheel, such primary

suspension spring comprises a transverse elastic beam which is spaced longitudinally from the control arms and thus may be in an area of the vehicle not required for other components. Substantial space is thus made available laterally between the control arms, for example for the drive aggregate, as well as the space near and above the road wheels normally consumed by compressible coil springs on MacPherson struts or otherwise.

This general form of arrangement of a transversely extending primary suspension spring spaced in the longitudinal direction of the vehicle from the control arms has been proposed in the past but appears to have been hitherto commercially unsatisfactory. Thus United States Patents 3,701,542 and 3,831,966 (Grosseau) each show a suspension arrangement including a transverse primary spring beam, conventional quadrilateral suspension linkage and torsion bars extending from the lower quadrilateral arms along their pivot axes to connect such control arms to the ends of the suspension spring. Such an arrangement necessitates that the bending modes of the transverse spring caused by normal vehicle ride and handling deflections of the road wheels can be accommodated only by specially proposed connecting systems between the ends of the transverse spring and the two torsion bars. Such proposals form the subject of the patents, and are considered to be clearly unsatisfactory from several functional and structural standpoints.

The vehicle suspension system in accordance with the present invention, on the other hand, represents a commercially superior arrangement of a transverse spring suspension.

It has been found in the course of development of the present invention that reinforced polymeric material such as fiberglass-filled polyester resin exhibits superior properties for use in a spring beam. The present arrangement of the transverse spring beam makes effective use of a beam made of such material not only as the primary suspension spring but also as the principal or only anti-roll element for control of vehicle handling characteristics. A practical arrangement is thereby provided which not only avoids elaborate prior-art types of interconnection devices between the deflecting wheel control arms and the suspension beam but which also eliminates functional deficiencies of the prior-art constructions.

The vehicle suspension system in accordance with the present invention features the use of interconnection elements, such as torsion bars, which are simply fixed each at one end to a control arm and aligned along longitudinal axes of the chassis, spaced outboard of the control arm pivot axes, and then simply fixed at their other ends to the transverse beam terminal ends in such fashion that con-

control arm deflections cause the transverse beam to experience bending modes which are characterized by an inboard pair of nodal points within the beam at fixed locations of the

5 sprung mass. This nodal point bending characteristic at once affords and is the result of the simple connections just described. With the Grosseau structures, the interconnections at the ends of the transverse beam, although
10 elaborate, are nevertheless subject to large amounts of relative sliding motion between the beam ends and the interconnecting element as beam deflection occurs, and the torsion rods are loaded in bending modes as
15 well as torsionally. With the present invention, the nodal-point bending characteristic permits geometry wherein no appreciable such relative displacement will occur.

Further, should long torsion rods be chosen
20 as interconnecting elements, as is the case in the preferred embodiment of the present system, the further advantage is derived from the fact that no longitudinal bending of the torsion rod occurs, and there need be no independent structural support of such rod on the
25 chassis sprung mass, in contrast to what is required in the prior art.

More specifically, the nodal-point beam bending characteristic of the present invention
30 contemplates that the spacing between the control arm axis and the torsion rod axis may be so selected with reference to the material properties of the beam and the torsion rod, if such be used, that both the beam-end and the
35 control-arm anchorages of the torsion rod ends describe generally identical arcuate paths in parallel planes in space during wheel deflection. The torsion rod, or other interconnecting device, acts simply to carry a pure
40 force couple between the control arm and the respective beam end. To complete the system, reaction load elements are provided on the chassis sprung mass at the nodal points of the
45 beam to serve as spring constraints during vehicle roll. The entire such system permits the use of lightweight polymeric spring already referred to as the sole or principal ride and anti-roll rate control element, without the
50 need for any undue sacrifice in one function to favor the other. Further advantage in this respect may be anticipated in that by judicious design of such nodal-point reaction elements, even further latitude will be available to the designer in the selection of the ride and
55 the anti-roll behaviour of the spring beam and thus of the vehicle.

The use of all the aforementioned realized objectives and features of this invention results in a suspension package of light weight,
60 high space utilization efficiency and simplicity of construction.

It is even possible for the spring beam to be easily contained within a relatively shallow box-section cross member of the vehicle,
65 where the spring is at a low position freeing

space for engine aggregates and yet is protected from road hazards.

The suspension may additionally feature design height adjustment provisions wherein the
70 torsion rods of the exemplary embodiment are connected to the respective control arms via adjustable lever arm devices capable of providing variable prestressing of the torsion rod and spring beam, for supporting the vehicle
75 sprung mass at variable design heights.

In the drawings:

Figure 1 is a fragmentary perspective view partially in phantom, of a vehicle chassis sprung mass having one embodiment of a
80 wheel suspension therefor in accordance with the present invention;

Figure 2 is an enlarged fragmentary elevation, partially in section, generally in the plane indicated by the line 2-2 of Fig. 1, in the
85 direction of the arrows, and illustrating the wheel suspension in a plurality of wheel positions;

Figure 3 is a reduced-size fragmentary section, with parts in elevation, generally in the
90 plane indicated by the line 3-3 of Fig. 2, in the direction of the arrows;

Figure 4 is a view similar to Fig. 3 but generally in the plane indicated by the line
4-4 of Fig. 3, in the direction of the arrows;
95 and

Figures 5 to 7 are diagrammatic representations of various bending modes of the spring beam of the suspension system.

Referring particularly now to Fig. 1 of the drawings, the same illustrates the front portion of an automotive vehicle chassis and front suspension, in accordance with this invention, for the steerable front road wheels as indicated in phantom at 10. Such road wheel
100 is adapted for connection via a drive shaft assembly 12 to a front drive aggregate which may be located in the front compartment of the vehicle body or sprung mass, all as well known in the art. The present invention is
105 directed to the packaging and body styling challenges arising with such front-drive design, but it should be noted that this invention is equally well adapted to vehicles with rear drive aggregates or otherwise where space
115 utilization efficiency is a problem.

In the front drive vehicle shown, the vehicle sprung mass or chassis/body may comprise a drive aggregate cradle 14 including a pre-assembled combination of stamped sheet
120 metal welded box section rails 16 and fore and aft spaced box section cross members only generally indicated at 18 and 20. Such cradle assemblies support the vehicle drive aggregate, not shown, and are conventionally
125 bolted to the remainder of the vehicle body via isolation mounts.

A vehicle wheel suspension in accordance with this invention is, in the instant preferred embodiment, structured to include members
130 presently found in vertical strut/shock suspen-

sions, but it should be understood that the principles of the invention are not thus limited. A vertical strut/shock 22 comprises an outer shock cylinder 24 and a telescoping piston rod 26 suitably anchored to an upper sheet metal tower structure 28 of the vehicle body sprung mass, as via a conventional isolation coupling indicated at 30 in Fig. 2. As well known, such strut/shock 22 may further include a saddle bracket 32 welded or otherwise fixed to cylinder 24 and including ears for reception of a spaced pair of through-bolt and nut assemblies 34 which attach the strut/shock 22 to an upper portion of the usual wheel support or knuckle 36. As is known, such knuckle, in the case of driven or live axle road wheel, includes provision for passage of the drive shaft 12 through the knuckle as a live spindle as here shown. The space utilization advantages of the invention, while generally directed to the problems of drive aggregate packaging, may equally apply to dead spindle suspensions remote from the drive aggregate, or to older forms of propeller shaft rear drive vehicles.

The suspension of this invention includes for wheel 10 a transverse control arm 38 swingably mounted on the vehicle chassis sprung mass, as at rail 16. The control arm is thus mounted to swing on an axis A, at the center of an aligned pair of pivot assemblies which may be conventionally structured of rubber bushed sleeves 40 seated in spaced legs of the control arm and affixed to brackets 42 of the rail 16 via bolt/nut fasteners 44, Figs. 3 and 4. At the outboard end of such control arm 38 there is provided a conventional ball joint 46 installed within a lower portion of knuckle 36. As is known, such ball joint cooperates with strut/shock 22 to define a steer axis for steerable road wheel 10 under actuation of conventional steering linkage, not shown.

Referring again to Fig. 1, in accordance with the principles of the invention, there is provided a transverse spring beam 48. The spring is contained within the confines of the front box section cross member 20 and extends completely therethrough to have symmetry about the vehicle centerline and identical connection with the opposite steerable road wheel 10 in the same manner as to that now to be described for the road wheel 10 shown. Thus, the descriptions given above and hereinafter to the structure illustrated are identically applicable to the other mirror half of the instant suspension embodiment. As to beam 48, while the invention is in no way limited to a selection of spring material it has nevertheless been found that the invention is best embodied with a beam constructed of reinforced polymeric material such as fiberglass reinforced polyester resin. Such a spring beam achieves light weight while still absorbing high amounts of strain energy, and ex-

hibits a modulus of elasticity well adapted to use for both ride and vehicle roll rate control as applied in the manner now to be described.

As seen in Figs 2 through 4, the spring beam 48, in accordance with this invention, is arranged to have its terminal ends extend substantially laterally beyond vertical planes B containing the swing axes A of the opposite control arms 38. In the instant embodiment, control arms 38 are connected to those extended ends of spring beam 48 by a pair of torsion rods 50. The torsion rods each lie along an axis C, Figs. 3 and 4, which lies outboard of the swing axes A of the control arms. Each terminal end of the spring beam 48, indicated by the numeral 52, carries a clamp assembly 54 comprised of mated upper and lower sections enveloping the end of the beam and joined by fasteners 56 extending through apertures in the beam end 52, and further fasteners 58. A rubber or like material isolation sleeve 60 intervenes beam end 52 and the clamp assembly. The upper part of the clamp assembly includes a boss 62 suitably apertured and provided therein with internal splines mated with external splines on the forward ends of torsion rods 50, thereby to establish torque-carrying connection of each torsion rod to the spring beam 48 at axis C.

At its other end, the torsion rod carries similar external splines and is received within a design height adjustment device for control arm 38. Such device comprises an elongated tubular lever member 64 having its rearward end rotatably carried within a pocket of the control arm as shown in Figs. 3 and 4. The lever member has an arm or crank portion 66 extending transversely toward the ball joint 46. As seen best in Fig. 2, the end of such crank portion is apertured and threaded to receive the threaded shank of a design height adjusting stud 68 which projects through an aperture of the control arm 38 and rests upon a beveled bolt head seat 70 fixed thereon. Thus, the torsion rods 50 are placed in torque-carrying relationship between the control arms, which swing about axes A, and the elastic spring beam 48. As seen in Fig. 5, with the front end weight of the vehicle sprung mass imposed in shares $W/2$ upon the bushings 40 at each axis A, the ground reaction $W/2$ on each road wheel 10 creates a force couple M within each control arm 38 representative of vehicle weight only, and tending toward counterclockwise rotation of the control arm viewed in Fig. 2. Such force couple M is carried via the torsion rods 50 at axes C to the beam ends 52 thereby to cause the beam 48 to be stressed and deflected in accordance with its modulus of elasticity and torsion rods 50 are torsionally deflected in accordance with their elasticity. The result is a specific angular position of control arm 38 defining a so-called "design height" for a

particular sprung mass at so-called "design loading", e.g., 2 passengers and a quantity of cargo. The lever member 64 provides a convenient adjustment of the position of each control arm 38 about swing axis A in the presence of such deflections under the weight of that particular sprung mass. The design height of the latter is simply adjusted by threaded rotation of stud 68 to vary the angular relationship of the control arm, and thus the sprung mass, in relation to the crank portion 66 of the loaded torsion rod 50.

The result of the interassociation of parts thus far described is to cause spring beam 48 to have defined therein a spaced pair of neutral or nodal deflection points as at axis D, Figs. 1, 2 and 4. Thus, as viewed in Fig. 2, dynamic road forces causing deflection of road wheel 10 in an upward direction to the broken line position shown, causes the torsion rod to impose an increased force couple at terminal beam end 52 in such a manner as to bend the spring beam to a condition such as indicated at 48'. This bending occurs with the beam remaining essentially vibratorily undisturbed in space or relative to the sprung mass cross member 20 at the location of nodal axes D. Such is also true for an opposite deflection of the road wheel 10 and deflection of spring beam 48 to a position as indicated at 48'' in Fig. 2. This beam bending nodal point characteristic is true whether the opposite front road wheels 10 are deflected in the same direction, as during pure ride motion, or during a turning maneuver where a roll couple imposes forces producing opposite wheel deflections across the car. As seen in Fig. 6, in the former case of pure ride motion, the beam is in "pure bending" as the term is known, i.e. equal force couples ΔM alone acting counter-rotatively on the beam ends, and with no vertical reaction forces required to hold the beam fixed in space or relative the sprung mass. As seen in Fig. 7, during roll motion, the bending is converted to some higher order sinuous curvature to accommodate within the beam the situation where the force couples are of the same sense tending to displace the beam bodily in space relative the sprung mass. In such case vertical reaction forces F are required in space and the same are provided characteristically of this invention by appropriate spring reaction mounts, to be described, located at the spaced pair of nodal points at axes D. A variety of dynamic road force conditions may of course be induced in the suspension combining some part of each of the two theoretically pure cases shown in Figs. 6 and 7, but the beam 48 is expected to bend about its characteristic nodal points at axes D in any case.

It is a feature of such beam bending characteristic in any of these cases that all such bending may proceed with no significant relative motion between the beam ends 52 and

the respective control arm 38 at either side of the vehicle. Thus, as viewed in Fig. 2, the point on control arm 38 in some transverse vertical plane therethrough wherein torsion rod 50 is centered can describe virtually the same arc of travel as will the point on the beam end 52 of the spring beam in a parallel transverse vertical plane therethrough where the other end of torsion rod 50 is centered. These are identified by the single arc C at each beam end appearing in Figs. 6 and 7. The curved paths of the beam ends 53 are essentially described around the nodal axes D, while the circular arcs of travel of the rearward end of the torsion rod 50 are of course described about axes A of the bushings 40. Some minute difference in curvature may exist, but by appropriate selection of the elasticity of the torsion rods and of spring beam 48, these paths in the two spaced transverse planes may be essentially such that there need be no bending forces of any importance longitudinally along axes C in the torsion rods 50. Such torsion rods thus simply carry the pure force couples associated with deflections of road wheels 10 about axes A. There need be no independent support of the torsion rods on the sprung mass. And the location of torsion rods 50 away from axes A inherently simplifies the construction of control arms 38 and the bushings 40 over prior practice in which torsion rods are placed directly on such axes A and the control arm pivots must accordingly be specially modified. As seen best in Fig. 2, the reaction mounts at axes D for the system are indicated at 72 and each simply comprises an opposed pair of rubber or like material tapered blocks lightly force-fitted between interior surfaces of cross member 18 and beam 48. The hardness of the rubber is selected to best achieve isolation of road disturbances from the vehicle sprung mass without significantly softening within the free-body force situations represented in Fig. 7, the vertical restraint of nodal axes D under forces F. It is expected, however, that the shape of the blocks, or the material softness thereof, or some appropriate combination of selections thereof may very well prove to impart further advantage to the instant invention in allowing yet another area of design flexibility wherein the roll rate of the suspension, for example, might be readily adjusted without affecting ride rate, simply by alterations in the behaviour of spring beam 48 at its nodal points.

Due to the lateral spacing between rotational axes, so as to speak, of the terminal beam ends 52 at D and the control arms at A, there is a consequent mismatch of the angular deflection which the two ends of torsion rod 50 undergo during wheel deflection. Such mismatch induces torsional deflection in such rods 50 during the wheel deflection and gives rise to the consideration of the material of the

torsion rod, also for appropriate design of the ride and roll characteristic of the suspension system. As indicated hereinabove, the invention is however not limited to the use of actual such torsion rods inasmuch as other means, such as longitudinal extensions of stamped metal form control arms 38, for example, may provide acceptable results. A variety of concerns such as cost, weight, and the desired location of the transverse spring beam longitudinally apart from the control arms for efficient space utilization, will all enter in making such choices. It is manifest that the principles of the invention give rise to a number of such possibilities enabling practical suspension constructions of the instant general type not heretofore available.

Claims:

1. A suspension system for the sprung mass of a vehicle relative to a transversely opposite pair of road wheels, comprising a pair of transverse control arms each adapted to carry a respective one of the wheels and swingably mounted at a respective side of the sprung mass on a generally longitudinally extending axis thereof, a primary suspension spring comprising an elastic beam extending transversely of the sprung mass proximate the control arms but spaced from the control arms in a direction longitudinally of the sprung mass and having end portions projecting outboard of generally vertical planes containing the swing axes of the control arms, means connecting each control arm to a respective end portion of the elastic beam at a location on the latter outboard the respective said generally vertical plane and operative to carry to the beam forces effective to deflect the control arms relative to the sprung mass, whereby under the action of such forces the beam is adapted to bend about a spaced pair of nodal points located substantially inboard of the connection locations, and reaction mount means on the sprung mass engaging the elastic beam at the nodal points thereof.

2. A suspension system according to claim 1, in which the primary suspension spring comprises a beam of elastic polymer material, and torsionally elastic means connect each control arm to a respective end portion of the beam.

3. A suspension system according to claim 1 or 2, in a vehicle having a frame cross-member spaced longitudinally from the transversely opposite pair of road wheels, in which the elastic beam is disposed within the frame cross-member.

4. A suspension system according to any one of claims 1 to 3, in which the reaction mount means on the sprung mass and engaging the beam at the nodal points thereof comprises an opposed pair of elastomer blocks seated on the cross-member.

5. A suspension system according to any

one of claims 1 to 4, in which the means connecting the control arms to the respective end portions of the beam comprise a pair of torsion rods extending on axes generally parallel to but outboard of the swing axes of the control arms and each connected at one end to a respective control arm and affixed at the other end to an end of the beam.

6. A suspension system according to any one of claims 1 to 5, in which the control arms are each connected at the said one end thereof to the respective control arm in an adjustable fashion by way of respective sprung mass height adjustment means.

7. A suspension system for the sprung mass of a vehicle relative to a transversely opposite pair of road wheels, substantially as hereinbefore particularly described and as shown in the accompanying drawings.

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